

Optimum Heating of Thick-Walled Pressure Components Assuming a Quasi-steady State of Temperature Distribution

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As a result of the development of wind farms, the gas – steam blocks, which shall quickly ensure energy supply in case the wind velocity is too low, are introduced to the energy system. To shorten the start-up time of the gas – steam and conventional blocks, the structure of the basic components of the blocks are changed, e.g. by reducing the diameter of the boiler, the thickness of its wall is also reduced. The attempts were also made to revise the currently binding TRD 301 regulations, replacing them by the EN 12952-3 European Standard, to reduce the allowable heating and cooling rates of thick walled boiler components. The basic assumption, on which the boiler regulations allowing to calculate the allowable temperature change rates of pressure components were based, was the quasi – steady state of the temperature field in the simple shaped component, such as a slab, cylindrical or spherical wall.

Keywords: pressure component, thermal stress, quasi-steady state, optimum heating

Introduction

The quasi – steady state occurs in structural components of boilers and turbines during a long heating or cooling process running at a constant rate of change of temperature. In this paper, a more detailed analysis of the quasi – steady state was conducted for non – weakened walls and for walls weakened by openings. The use of formulas for circumferential stress at the edges of the openings to determine the allowable heating and cooling rates for the assumed pressure was also presented. Those formulas can be used in real life, as the analysis of the temperature and stress fields in the quasi – steady state can be quickly conducted with the use of the Finite Elements Method (FEM), even for components of a complex structure.

The quasi – steady state can be characterized by the

following factors:

- constant temperature difference between the selected points of the component (Fig. 1)
- constant, time independent, thermal stress (Fig. 2).

The results presented in Figs. 1 and 2 were obtained for the following data: $r_m = 0.85$ m, $r_o = 0.94$ m, $c = 511$ J/(kg·K), $\rho = 7775$ kg/m³, $k = 47.3$ W/(m·K), $E = 201105$ MPa, $\beta = 1.21 \times 10^{-5}$ 1/K, $\nu = 0.288$.

Let's consider the simplifying assumption that the thermo-physical properties of the wall material: k , c , ρ , E , and ν are constant and determined for the average temperature over time and the wall thickness. The transient heat transfer equation in the quasi – steady state can be then simplified to form [1, 2]

$$\nabla^2 T = \frac{\nu_T}{\kappa} \quad (1)$$

Nomenclature

c	specific heat of the metal, J/(kg·K)	t	time, s
d_{in}	inner diameter of the drum, m	T	temperature, °C or K

d_{wo}	inner diameter of the downcomer, m
E	modulus of elasticity, MPa
h	heat transfer coefficient, $W/(m^2 \cdot K)$
k	thermal conductivity, $W/(m \cdot K)$
p	absolute pressure in the drum, MPa
p_o	atmospheric pressure, MPa
p_n	gauge pressure in the drum $p_n = p - p_o$, MPa
p_{nd}	design gauge pressure in the drum, MPa
r_{in}	inner radius of the drum, m
r_o	outer radius of the drum, m
\mathbf{r}	position vector
s	drum wall thickness, m
s_o	downcomer wall thickness, m
S	circumferential thermal stress at the downcomer edge, MPa

Greek Symbols

α_m	stress concentration factor for circumferential stress caused by pressure
α_T	stress concentration factor for circumferential thermal stress
β	linear thermal expansion coefficient, $1/K$
κ	thermal diffusivity, m^2/s
ν	Poisson's ratio
ρ	density, kg/m^3
σ	normal stress, MPa
ϕ	shape factor
ω	ratio of outer to inner diameter

Subscripts

a	allowable
f	fluid
0	initial

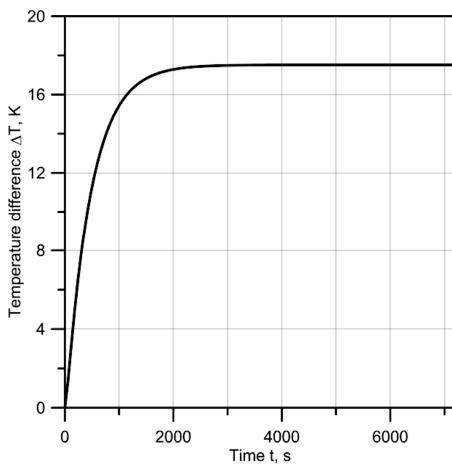


Fig. 1 The temperature difference between the internal and the external surface of the cylinder during heating at a constant rate $v_T = 3 \text{ K/min}$.

where $v_T = \frac{\partial T}{\partial t} = \text{const}$ is the rate of wall temperature change. The symbol ∇^2 denotes the Laplace operator. The thickness of the wall is constant, and the external surface $r = r_o$ of the pressure component is thermally insulated

$$\left. \frac{dT}{dr} \right|_{r=r_o} = 0 \quad (2)$$

The temperature on the internal surface $r = r_{in}$ rises linearly in time at a rate of v_T

$$T \Big|_{r=r_{in}} = T_{in} = v_T t \quad (3)$$

Expressing the Laplace operator in a cylindrical or

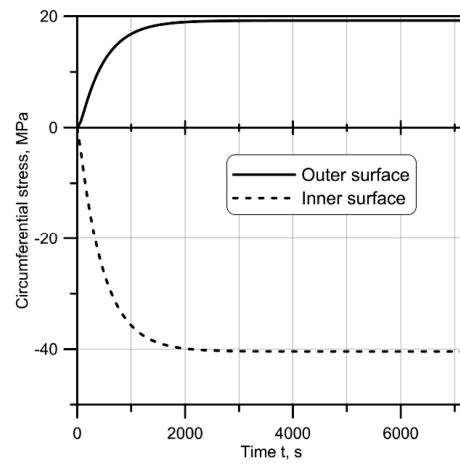


Fig. 2 Circumferential stress on the inner and outer surface of the boiler drum during heating at a constant rate: $v_T = 3 \text{ K/min}$.

spherical coordinate system leads to the specific form of the Eq. (1)

$$\frac{1}{r^m} \frac{d}{dr} \left(r^m \frac{dT}{dr} \right) = \frac{v_T}{\kappa} \quad (4)$$

where $m = 1$ for the cylindrical component and $m = 2$ for the spherical component. After double integration of the above equation and after determining the constants from the boundary conditions (2) – (3) for the cylindrical wall, the following expression is obtained

$$T = T_{in} + \frac{v_T}{4\kappa} \left(r^2 - r_{in}^2 - 2r_o^2 \ln \frac{r}{r_{in}} \right) \quad (5)$$

The average temperature over the wall thickness is

calculated from the formula

$$T_m = \frac{m+1}{r_o^{m+1} - r_{in}^{m+1}} \int_{r_{in}}^{r_o} r^m T dr \quad (6)$$

Substituting Eq. (5) into Eq. (6) with $m = 1$ and integrating gives the average temperature T_m on the thickness of the cylindrical wall

$$T_m = v_T t + \frac{v_T s^2}{\kappa} \frac{(u^2 - 1)(3u^2 - 1) - 4u^4 \ln u}{8(u^2 - 1)(u - 1)^2} \quad (7)$$

where: $u = r_o / r_{in}$ - the ratio of the outer radius to the inner radius, $s = r_o - r_{in}$ - wall thickness.

The circumferential thermal stress σ_{ww} on the inner surface is calculated as follows

$$\sigma_{ww} = \frac{E\beta}{1-\nu} \frac{v_T s^2}{\kappa} \phi_{ww} \quad (8)$$

The circumferential thermal stress on the outer surface is calculated from the similar formula

$$\sigma_{wz} = \frac{E\beta}{1-\nu} \frac{v_T s^2}{\kappa} \phi_{wz} \quad (9)$$

where the so-called shape factors for the internal surface ϕ_{ww} and the external surface ϕ_{wz} are given by the formulas

$$\phi_{ww} = \frac{1}{8} \frac{(u^2 - 1)(3u^2 - 1) - 4u^4 \ln u}{(u^2 - 1)(u - 1)^2} \quad (10)$$

$$\phi_{wz} = \frac{1}{8} \frac{(u^4 - 1) - 4u^2 \ln u}{(u^2 - 1)(u - 1)^2} \quad (11)$$

Similarly, the average temperature over the thickness of the sphere can be derived from Eq. (6). The shape factors for a sphere: ϕ_{kw} for the inner surface and ϕ_{kz} for the external surface, are given by

$$\phi_{kw} = \frac{1}{15} \frac{9u^5 - 5u^6 - 5u^3 + 1}{(u^3 - 1)(u - 1)^2} \quad (12)$$

$$\phi_{kz} = \frac{1}{10} \frac{(u^5 - 1) - 5u^2(u - 1)}{(u^3 - 1)(u - 1)^2} \quad (13)$$

The shape factors at the inner and outer surface of cylindrical and spherical walls are presented in Fig. 3.

The absolute values of the shape factors for the spherical wall ϕ_{kw} and ϕ_{kz} are smaller than the appropriate values of the shape factors for the cylindrical wall ϕ_{ww} or ϕ_{wz} (Fig. 3).

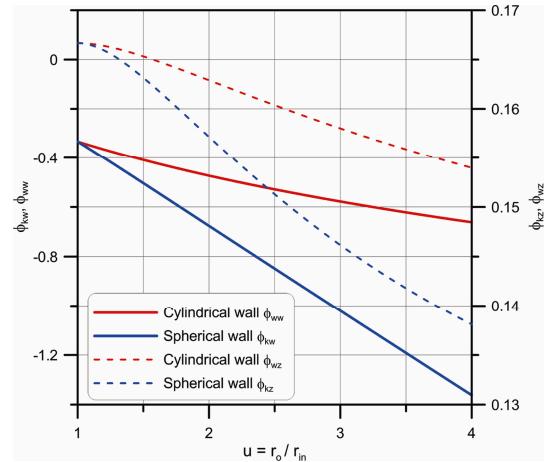


Fig. 3 Shape factors for the inner and outer surface of cylindrical and spherical walls.

For the same rate of temperature changes, the thermal stresses in the spherical walls will be greater than the thermal stresses in the cylindrical walls.

Optimum Heating of Components Without Openings

Heating or cooling of the components is conducted in a way, which will assure that the circumferential thermal stress on the inner surface equals the allowable stress.

$$\sigma_{ww} = \sigma_a \quad (14)$$

Substituting Eq. (8) into Eq. (14) gives

$$\frac{E\beta}{1-\nu} \frac{v_T s^2}{\kappa} \phi_{ww} = \sigma_a \quad (15)$$

After transformation of Eq. (15), the expressions for the heating or cooling rate of the thick walled, cylindrical or spherical boiler components are obtained

cylindrical wall

$$v_T = \frac{\sigma_a \kappa}{s^2 \phi_{ww} \frac{E\beta}{1-\nu}} \quad (16)$$

spherical wall

$$v_T = \frac{\sigma_a \kappa}{s^2 \phi_{kw} \frac{E\beta}{1-\nu}} \quad (17)$$

The temperature change rate is positive during the heating process and negative during the cooling process. The wall temperature depends on the fluid temperature, but the temperature difference between the selected points in the components is constant and independent of time and the value of the fluid temperature.

Quasi – Steady Distributions of the Temperature and Stresses in Pressure Components with Openings

The quasi - steady state temperature distribution in components of complex shape can be determined using Eq. (1) with the appropriate boundary conditions. Equa-

tion (1) describes a steady state temperature distribution in the body $T(r,t)$ with uniformly distributed heat sinks with the specific power equal to: $\dot{q}_s = v_T / \kappa$.

In the following, an application of the quasi – steady state approach for determining allowable heating rates in components with complex shape will be discussed. Since stresses at points P_1 and P_2 (Fig. 4) limit the heating rate of the boiler drum or other thick-walled vessels weakened by nozzles [3-6], thermal and pressure stresses are calculated for these points.

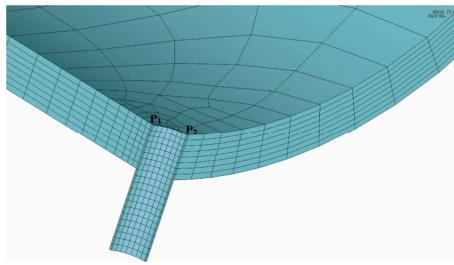


Fig. 4 Section of the boiler drum – downcomer junction and mesh of finite elements used for thermo-mechanical analysis of the drum – downcomer junction

The following geometrical dimensions of the boiler drum: inner diameter of the drum $D = 1.7\text{m}$, inner diameter of the downcomer $d = 0.09\text{m}$, thickness of the drum wall $H = 0.09\text{ m}$, thickness of the downcomer wall $h = 0.006\text{ m}$ and physical properties of the boiler material: $k = 42\text{ W/(m}\cdot\text{K)}$, $c = 538.5\text{ J/(kg}\cdot\text{K)}$, $\rho = 7800\text{ kg/m}^3$, $E = 1.96 \times 10^{11}\text{ N/m}^2$, $\beta = 1.32 \times 10^{-5}\text{ 1/K}$ and $\nu = 0.3$ were assumed for the calculations. At first, time changes of the circumferential stress in time at point P_1 for different heat transfer coefficients were computed (Fig. 5). An analysis of the results shows that the quasi – steady state at low values of the heat transfer coefficient $h = 500\text{W}/(\text{m}^2\cdot\text{K})$ is set only after a long heating time at a constant rate. After 4000 seconds of heating the stresses did not reach the constant values, characteristic for the quasi – steady state. Equivalent stress distributions in the boiler drum – downcomer junction determined using ANSYS for different pressures and heating rates are depicted in Figures 6, 7, and 8. The largest stress due to the pressure is at the point P_1 which is located at the hole edge (Fig. 6).

The equivalent stress at the point P_1 is about five times larger than the stress at the point P_2 when the drum is loaded only by the inner pressure. If the pressure inside the drum is equal to zero, then the equivalent thermal stress at the point P_2 is higher in comparison with the stress at the point P_1 (Figs 7a and 7b). When pressure is increased to design pressure $p = 10.87\text{ MPa}$ then the stresses due to pressure for $v_T = 1\text{ K/min}$ at the point P_1

exceed thermal stresses and the largest equivalent stress occurs at the point P_1 (Fig.8a). After increasing the heating rate to $v_T = 12\text{ K/min}$ negative thermal stresses exceed positive stresses due to pressure and the largest equivalent stress is at point P_2 (Fig. 8b).

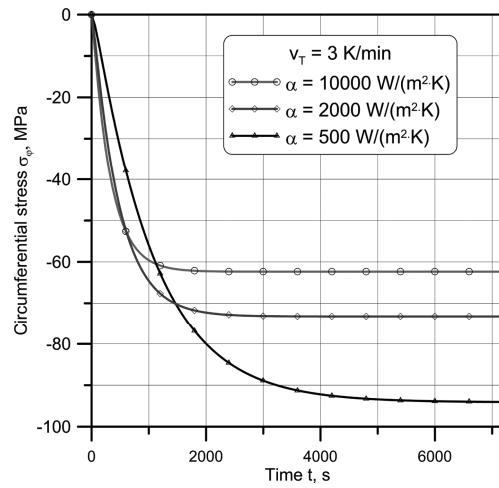


Fig. 5 Time variations of the circumferential stress σ_ϕ in time at point P_1 for different heat transfer coefficients h .

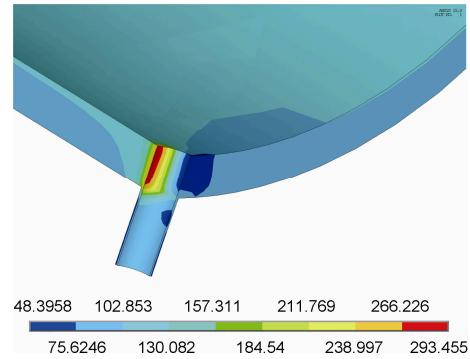


Fig. 6 Equivalent stress distribution caused by inner pressure

The stress concentration coefficients α_T at the point P_1 and P_2 (Fig. 4), for the quasi – steady state as a function of the heat transfer coefficient h are presented in Fig. 9. Thermal stresses determined from Eqs (8) and (9) refer to simple shapes, such as hollow cylinders and spheres. In real life, components used in power units often have openings, which can be viewed as material discontinuity causing stress concentration, what shall be taken into consideration when determining the allowable heating or cooling rate. Most often the assumed value of the thermal stress concentration coefficient at the edge of the opening is: $\alpha_T = 2$. Then, the influence of the heat transfer coefficient h on the thermal stress concentration coefficient α_T at the edge of the opening is not considered.

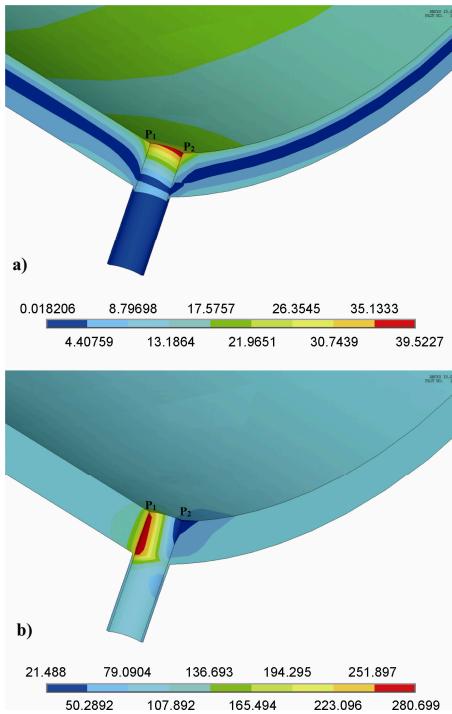


Fig. 7 Equivalent stress distributions at the drum – downcomer junction without pressure load (gauge pressure $p = 0 \text{ MPa}$) for different heating rates; a) $v_T = 1 \text{ K/min}$, b) $v_T = 12 \text{ K/min}$

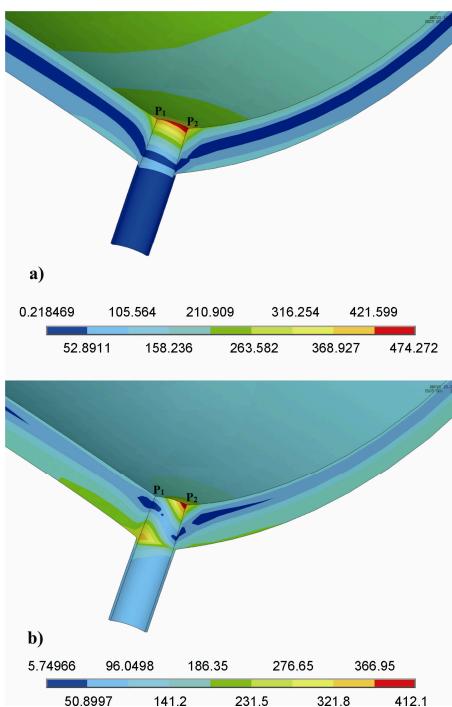


Fig. 8 Equivalent stress distributions at the drum – downcomer junction without pressure load ($p = 10.87 \text{ MPa}$) for different heating rates; a) $v_T = 1 \text{ K/min}$, b) $v_T = 12 \text{ K/min}$

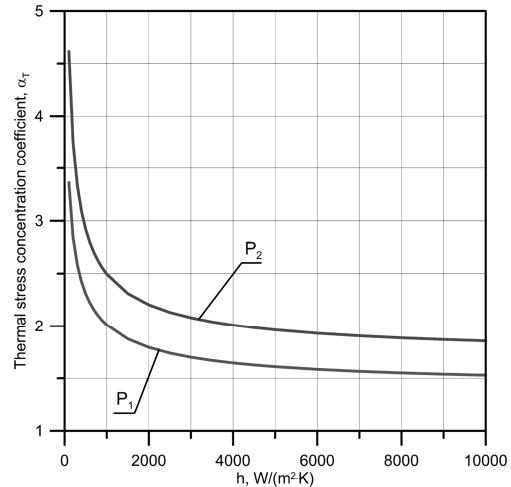


Fig. 9 Thermal stress concentration coefficient α_T as a function of the heat transfer coefficient h

However, the thermal stress concentration coefficient at the opening edge in the complex shaped components depends on the heat transfer coefficient h . Since the boiler drum has openings for downcomers, an intense concentration of stress is present at the edge of the opening. It can also be noted that the stress concentration coefficient at the edge of the opening strongly influences the heat transfer coefficient on the inner surface of the boiler drum and the downcomer. The value of the stress concentration coefficient α_T is lower at point P_1 – as compared to its value at point P_2 – regardless of the value of the heat transfer coefficient (Fig. 9).

When the heat transfer coefficient is equal to: $h = 10000 \text{ W/m}^2\text{K}$, the stress concentration coefficient at point P_1 is: $\alpha_T = 1.53$, and at point P_2 : $\alpha_T = 1.86$. The thermal stress concentration coefficient was calculated using the formula

$$\alpha_T = \frac{\sigma_{\phi,P_1}}{\sigma_{\phi,w}} \quad (18)$$

where σ_{ϕ,P_1} denotes circumferential stress at the edge of the opening at point P_1 , and $\sigma_{\phi,w} = \sigma_{\phi}(r_w)$ stands for circumferential stress on the inner surface of the boiler drum or pressure vessel without openings.

The total circumferential and equivalent stress at the edge of the opening at points P_1 and P_2 can be approximated linearly as a function of the heating rate. At the beginning of the boiler start – up, the gauge pressure in the drum is equal to zero thus the absolute values of circumferential and equivalent stresses at the points P_1 and P_2 are equal (Fig. 10). The total circumferential and equivalent stresses presented in Fig. 11 were computed considering pressure and thermal loads.

The circumferential stress at points P_1 and P_2 for pres-

sure $p = 10.87$ MPa can be approximated by a linear function with respect to v_T

$$\sigma_\phi = a + b v_T \quad (19)$$

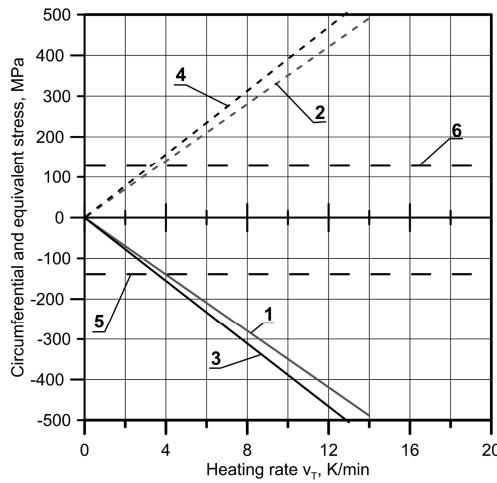


Fig. 10 Circumferential stress and equivalent stress at the edge of the downcomer opening at points P_1 and P_2 for the gauge pressure $p = 0$ MPa; 1 – circumferential stress at point P_1 , 2 – equivalent stress at point P_1 , 3 – circumferential stress at point P_2 , 4 – equivalent stress at point P_2 , 5 – allowable circumferential stress, 6 – allowable equivalent stress

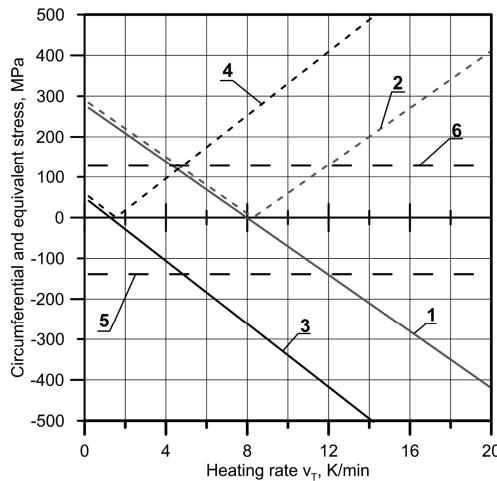


Fig. 11 Circumferential stress and equivalent stress at the edge of the downcomer opening at points P_1 and P_2 for the pressure $p = 10.87$ MPa; 1 – circumferential stress at point P_1 , 2 – equivalent stress at point P_1 , 3 – circumferential stress at point P_2 , 4 – equivalent stress at point P_2 , 5 – allowable circumferential stress, 6 – allowable equivalent stress

The allowable v_{T1} and v_{T2} boiler drum heating rates can be determined from the condition $\sigma_\phi = \sigma_a$, where the circumferential stress σ_ϕ is given by Eq. (19). The same method can be applied to other components of complex

shapes. The issue of optimization of heating and cooling of thick walled boiler components was presented in [5] without the assumption of the quasi-steady state.

However, the developed method for determining the allowable rate of cooling or heating pressure components is simple and can be easily used in the practice.

Components with Non - uniform Circumferential Temperature Distribution

There is an uneven temperature distribution on the drum circumference due to the presence of the water and steam zone. During start-up of the boiler, steam condenses on the inner surface of the drum in the steam zone. During the steam condensation, the upper part of the drum is heated very quickly because the heat transfer coefficient at the time of condensation is very high. The bottom of the drum, occupied by the water, the coefficient of heat transfer from the water to the inner surface of the drum is low, and the heating of the lower part of the drum occurs much more slowly. Heating at different rates of the upper and lower portions of the drum results in a temperature difference between the top and bottom of the drum. It should be pointed out that the temperature of the upper part of the drum is higher than the temperature of the lower part. A similar phenomenon can be observed in the horizontal sections of the pipeline and superheater headers. At the beginning of the start-up of the boiler, pipes are cold and water vapor condenses on the walls of the pipes. At the bottom of the horizontal piping and other pressure components of the boiler water accumulates at the bottom, especially when the drainages of the elements are not functioning properly. Such a case occurs when the horizontal pipelines and superheater headers are plastically deformed. In a later stage of the boiler start-up superheated steam flows through the upper part of the pipeline while at the bottom of the pipeline is saturated water. This is the reason for the occurrence of very large temperature differences on the circumference of the pipe, which reach up to about 200K. For this reason, permanent deformation of pipes and headers often occur, which in turn makes that the draining horizontal elements are not working properly. The stresses caused by temperature non-uniformity on the circumference of the pipe may be higher than the stress caused by temperature difference across the thickness of the pipeline. In the current European Standard EN 12952-3, thermal stresses caused by the temperature difference on the circumference of the cylindrical element are not at all taken into account. In this paper, an analysis of the heating of horizontal thick walled components will be carried out taking into account the different values of heat transfer coefficients in the water and steam zone (Fig. 12). Calculations are performed assuming that the temperature of

the fluid increases linearly with time so that after some time forms a quasi-steady state temperature field. This paper examines thermal and strength loads of the drum of the steam boiler OP-210M during startup. Figure 13 illustrates the time changes of the water level. The water level in the drum situated $H_w = 10$ cm below the horizontal plane passing through the axis of the drum is taken as the zero level. The water temperature in the evaporator increases resulting in an increase in the specific volume of the water and rising water level in the drum. After about 12000 seconds, the excess water is discharged from the drum and the water level is lowered to a level close to zero. The time variations of the saturation pressure in the drum is depicted in Fig. 14. Measured time variations of the drum temperature at seven points shown in Fig. 14 indicate that temperature differences on the drum circumference, especially at the initial period of the drum heating are significant.

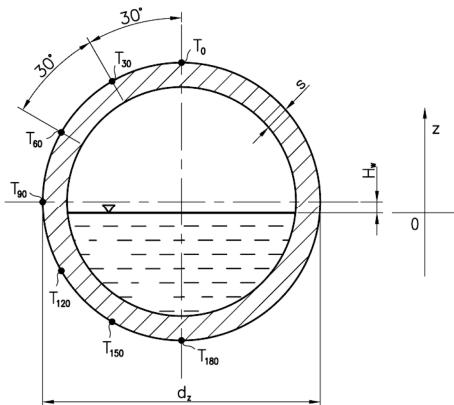


Fig. 12 The cross-section of the boiler drum with locations of thermocouples used for temperature measurement

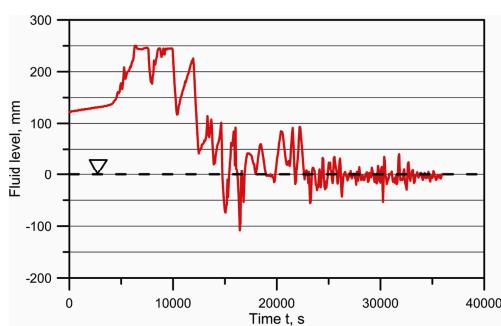


Fig. 13 Time variations of the level of water in the drum of OP-210M boiler during the start-up

As already explained earlier, the faster heating of the upper part of the drum is due to vapor condensation on the inner surface of the drum, during which the heat transfer coefficient is very high. After about 15000 seconds the temperature difference between the top and

the bottom of the drum is reduced considerably (Fig. 14). The maximum temperature difference between the top and bottom of the drum does not exceed 50 K. It must be emphasized that such a large temperature difference at the circumference of the drum is the cause of high thermal stresses. The analysis of the experimental results shows that in the initial stage of drum heating there are large temperature differences on the perimeter, which are the cause of additional thermal stresses. The current boiler standards do not take into account the boiler stresses due to temperature difference on the perimeter of the horizontal elements. Only quasi-stationary thermal stresses caused by the temperature difference across the wall thickness are considered.

Assuming the existence of quasi-stationary temperature field in the pressure element, the temperature difference on the component circumference caused by different heat transfer coefficients on the inner surface will be taken into account. The time variations of the circumferential stress at the edge of the drum – downcomer junction at the points P_1 and P_2 are shown in Figs. 15 and 16.

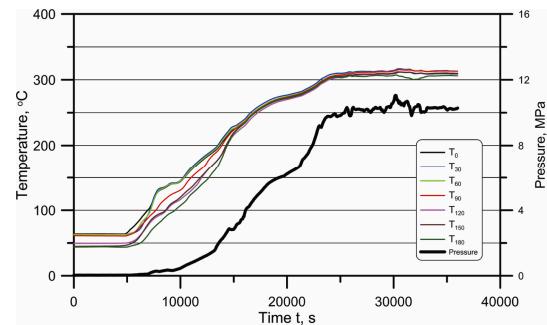


Fig. 14 Temperature and pressure variations on the drum circumference during the boiler start-up

The analysis of the results depicted in Figs. 15 and 16 shows that the absolute value of thermal stresses at the point P_2 is much greater than those at the point P_1 . It should be noted that the heat transfer coefficient in the water area is greater than $1000 \text{ W}/(\text{m}^2 \cdot \text{K})$. Due to the higher value of compressive thermal stress and a lower value of the tensile stress due to pressure occurring at point P_2 , the permissible rate of the drum heating should be determined due to the stress at the point P_2 . Changes in the circumferential stress and equivalent at points P_1 and P_2 as a function of heating rate are shown in Figure 17. In Figure 17a, the gauge pressure in the drum is $p = 0$, while in Figure 17b the pressure is equal to the design value of 10.87 MPa .

It should be emphasized that during the start-up of the boiler heat transfer coefficient in the water zone is less than in the steam zone. In such cases, a maximum allowable heating rate is determined by the stresses at the point P_1 .

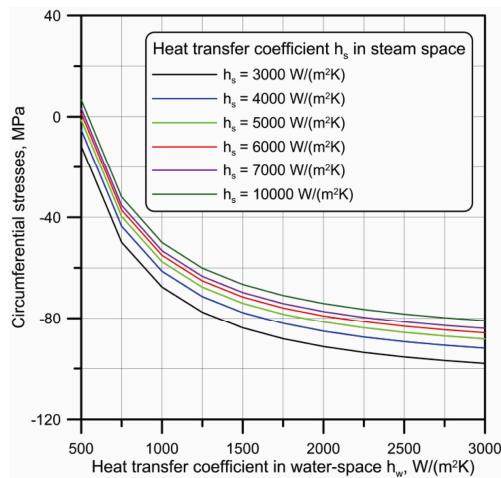


Fig. 15 Circumferential stresses on the hole edge at the point P_1

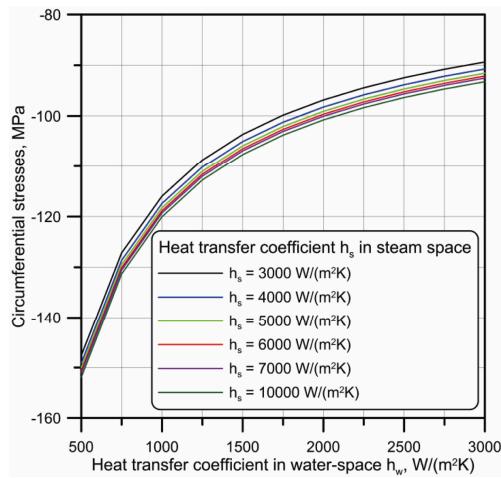


Fig. 16 Circumferential stresses on the hole edge at the point P_2

For a typical case that occurs in practice [7], when $h_w = 1000 \text{ W}/(\text{m}^2\text{K})$ and $h_s = 10000 \text{ W}/(\text{m}^2\text{K})$, the permissible heating rate at the beginning of the boiler start-up when $p = 0$ is approximately $3 \text{ K}/\text{min}$. Currently, the permissible rate of the drum heating recommended by boiler manufacturers is about $1 \text{ K}/\text{min}$. Allowable heating rate equal $3 \text{ K}/\text{min}$ (Figure 17a) resulting from the FEM analysis is significantly higher than the value prescribed by the boiler manufacturers. The start-up of the boiler can thus be shortened three times.

Conclusions

Summarizing the obtained results, it should be noted that the assumption of quasi – steady temperature field significantly simplifies the determination of the allowable rates of fluid temperature changes during the heating and cooling of boiler thick walled components. The

thermal stress concentration coefficients can be quickly determined with the use of the Finite Element Method (FEM), also for components of very complex shapes. For flat, cylindrical and spherical walls, in which the temperature distribution is one-dimensional, the thermal stress concentration coefficient does not depend on the value of the heat transfer coefficient. It was also shown that the heat stress concentration coefficient for the structural components of complex shapes depends on the value of the heat transfer coefficient on their inner surface.

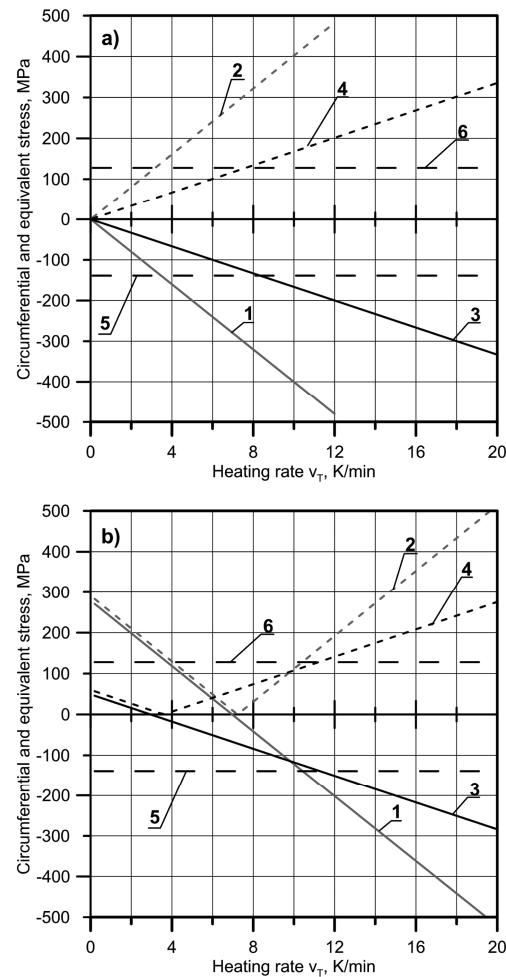


Fig. 17 Circumferential stress and equivalent stress at the edge of the downcomer opening at points P_1 and P_2 for $h_w = 1000 \text{ W}/(\text{m}^2\text{K})$ and $h_s = 10000 \text{ W}/(\text{m}^2\text{K})$; a) drum gauge pressure $p = 0 \text{ MPa}$, b) drum gauge pressure $p = 10.87 \text{ MPa}$; 1 – circumferential stress at point P_1 , 2 – equivalent stress at point P_1 , 3 – circumferential stress at point P_2 , 4 – equivalent stress at point P_2 , 5 – allowable circumferential stress, 6 – allowable equivalent stress

It has also analyzed quasi-steady state temperature field and stress in the drum – downcomer junction con-

sidering the different heat transfer coefficients in the water and steam space. Graphs presented in the chapter allow to determine heating rate limits at the beginning of the boiler start-up when the pressure is equal to 0 and at the end of starting when the pressure is equal to the nominal pressure. Permissible drum heating rates, assuming both circularly symmetric and asymmetric circular cross-section temperature field in the due drum obtained by the proposed method are several times higher than the limit values recommended by the manufacturers of boilers.

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